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LHC Interaction Region Quadrupole LQXA / LQXC Engineering Note for Complete Magnet Testing at Fermilab

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LHC Interaction Region Quadrupole LQXA / LQXC
Engineering Note for Complete Magnet Testing at Fermilab

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Chapter 1

LHC Interaction Region Quadrupole LQXA / LQXC

Engineering note for complete magnet testing at Fermilab

1.0 Introduction

This document constitutes the engineering note for the LHC interaction region quadrupoles being fabricated at Fermilab. It addresses the adequacy of the design and installation for testing single magnets at the Magnet Test Facility (MTF) within the Technical Division of Fermilab. Both generic and specific issues are addressed. Generic issues are those that pertain to the individual magnets themselves. Specific issues are those that apply only to the operating modes at Fermilab. For example, relief piping and relief valve analyses and discussions apply only to this specific installation, are not applicable to a string of magnets, and we make no attempt to generalize to that extent.

The magnet, piping and vacuum vessels will not be ASME Boiler and Pressure Vessel Code stamped vessels (hereinafter referred to as "the Code"). We do meet the Fermilab requirement to apply the design rules of the Code such that the intent of the Code is realized, i.e. that the geometry of all welds are consistent with the Code, that allowable stresses are met, etc. Fermilab manufacturing practices do not meet all of the Code requirements, most notably the continuous monitoring of all production processes, radiography of welds, etc. For that reason, Fermilab procedures require that allowable stresses be de-rated to 80% of their Code values. For the design and analysis of internal piping, we have applied the rules and practices outlined in ASME Code for Pressure Piping, B31.3, "Chemical Plant and Petroleum Refinery Piping". We have designed the bellows according to the standards of the Expansion Joint Manufacturers Association, Inc. (EJMA).

The chapters and appendices included in this note address each of the following major magnet systems in detail. Refer to the table of contents for the exact location of each analysis or component.

- Cold mass
- Internal piping
- Vacuum vessel
- Interconnects

1.1 Summary of results

It will be shown in each of the following chapters that the design of each magnet system is consistent with the operating requirements at MTF. Chapter 2 will address the cold mass in detail and will document a maximum allowable working pressure (MAWP) of 290 psi. The system relief settings at MTF are set at or below 100 psi. Chapter 3 will address the design of all internal piping and will show that it satisfies the requirements of ASME B31.3, when subject to the operating temperatures and pressures summarized in table 3.0.1. Chapter 4 documents the design and analysis of the vacuum vessel and shows that it meets the requirements of the Code as it applies to vacuum vessels and to section 5033 of the Fermilab ES&H manual when subject to all the applicable structural loads and the insulating vacuum load. Finally, chapter 5 documents the design and analyses of all interconnect bellows. The requirements, design rules, and calculation guidelines of the Expansion Joint Manufacturers Association (EJMA) were used throughout this chapter. EJMA is the recognized standards organization for bellows and is referred to throughout the ASME Boiler and Pressure Vessel Code.

We believe the designs of the systems documented in this note are adequate to ensure that their operation represents no hazard to personnel or to any of the external systems to which they will be connected.

Chapter 2

LHC Interaction Region Quadrupole

Q1 / Q3 Cold Mass Assembly

2.0 Introduction

The Q1 and Q3 cold mass assemblies in an LHC IR quadrupole consist of one KEK designed cold mass that forms the helium containment vessel. The cold mass consists of the following major components.

- Quadrupole collared coil assembly
- Cold iron yoke
- Outer helium containment vessel

The helium containment vessel consists of the following.

- One KEK designed cold mass (supplied by KEK)
- Two 304 stainless steel end dome assemblies
- One 316LN stainless steel beam tube

The purpose of the cold mass assembly is to maintain the collared coil assembly at its nominal operating temperature of 1.9 K and to act as the transport mechanism for liquid helium between magnets when they are installed at CERN. Under normal operating conditions, the temperature of the vessel is 1.9 K with an internal pressure of 4.4 psig [1.3 bar].

The cold mass must satisfy all the requirements of the "Pressure Vessels" section (section 5031) of the Fermilab ES&H Manual. This section states that all applicable vessels shall adhere to the requirements of the ASME Boiler and Pressure Vessel Code Section, VIII.

This vessel will not be an ASME code stamped vessel. The intent of the design is to address and adhere to as many requirements of the ASME code as possible.

The assembly can be seen in Figure 2.0.1.

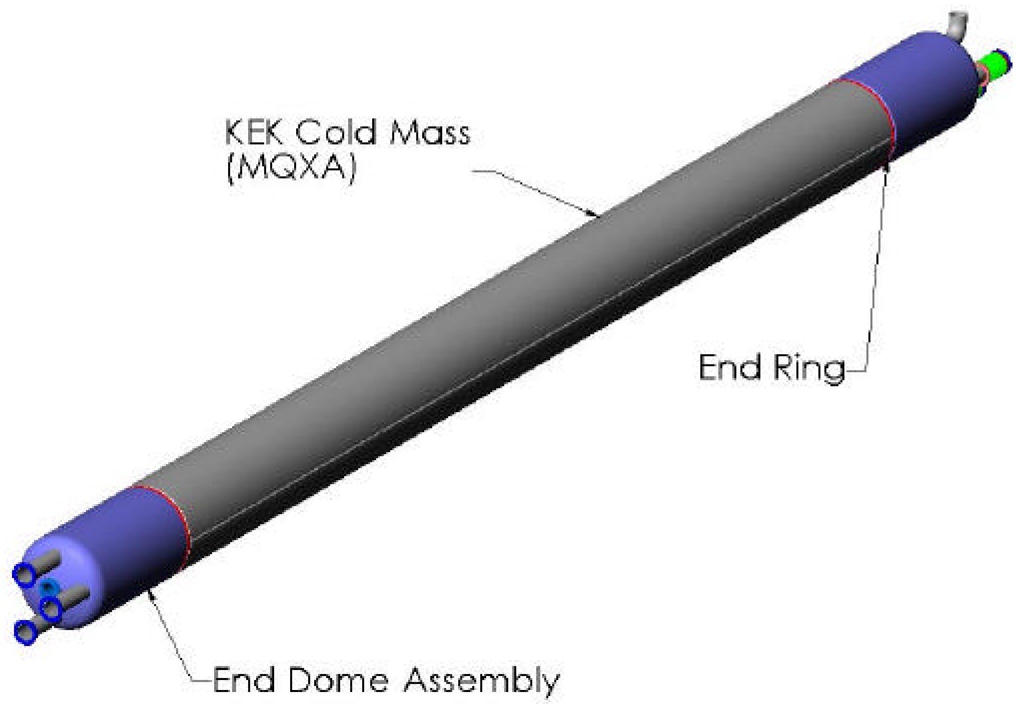

Figure 2.0.1 Cold Mass Assembly

The maximum stress that is allowed by Section II, Part D, Table 1A of the Code is as follows:

304 stainless steel: 20,000 psi

304L stainless steel: 16,700 psi

316LN stainless steel: 20,000 psi.

Section 5031 of the Fermilab ES&H Manual requires de-rating of the allowable stress to 80% of the allowed value in cases where the vessel is either fabricated in-house or is not code-stamped. This reduces the allowed stress in pressure vessel calculations to the following:

304 stainless steel: 16,000 psi

304L stainless steel: 13,360 psi

316LN stainless steel: 16,000 psi.

The design pressure for the LHC IR quadrupoles for CERN is 290 psi. This design pressure is the MAWP of the cold mass assembly. Should a quench occur on the test stand, there is no risk of over-pressurizing the cold mass since the feedbox at MTF is rated for 100 psi and has a relief set at or below this value.

2.1 Cold mass

The cold mass for the Q1 and Q3 magnet assemblies is designed at KEK and built by Toshiba. KEK is responsible to insure that the cold mass meets the ASME Code. Appendix A includes the engineering note that is prepared by KEK. KEK also performs a pressure test on the cold mass prior to shipment to FNAL.

The remainder of the helium containment vessel is the responsibility of FNAL. The additional parts are detailed below.

2.2 End dome Assembly

The end dome assembly is attached to both ends of the cold mass to create the helium containment vessel. There are pipes attached to openings in the dome as shown in Figure 2.2.1. These pipes transport helium as well as provide a feedthrough for the wiring between the magnet and the feedbox when installed at MTF.

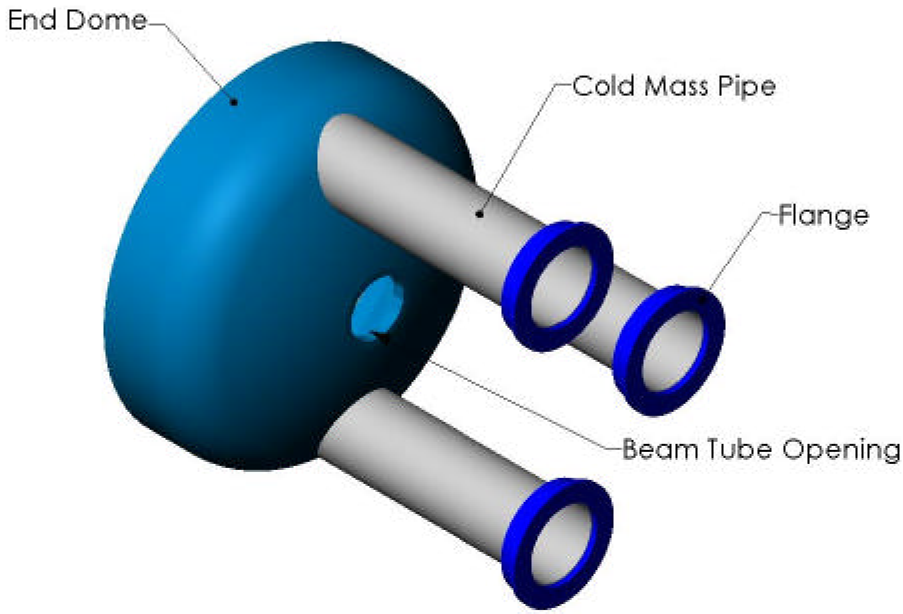

Figure 2.2.1 End Dome Assembly

2.2.1 End dome

The end dome is a formed ellipsoidal head. The minimum required thickness is given by UG-32 (d)

$$t = \frac{PD}{2SE - 0.2P}$$

where:

t = minimum required thickness of head after forming, inches

P = internal design pressure = 290 psi

D = inside length of the major axis (ID) = 18.504 inches

S = allowable material stress = 16,000 psi

E = joint efficiency = 0.60

Calculating:

t = [290(18.504)]/[(2*16000*.06)-0.2*290] = 0.280 inches

The head thickness is 0.687 inches so this requirement is satisfied.

This is the minimum required thickness for the end dome without any openings. Since there are openings in the dome, the requirement for reinforcement must be checked. There are four openings in the dome so the requirement for reinforcement is given by UG-42 of the Code, "Reinforcement of Multiple Openings". The center opening is for the beam tube. The other three openings are for the cold mass pipes. When considering the required reinforcement, Section UG-42 (a) (3) states "A series of openings all on the same center line shall be treated as successive pairs of openings." From this statement and the symmetry in the hole pattern, only two adjacent openings need to be addressed. These are the center opening and one of the three pipe openings. See Figure 2.2.1.1.

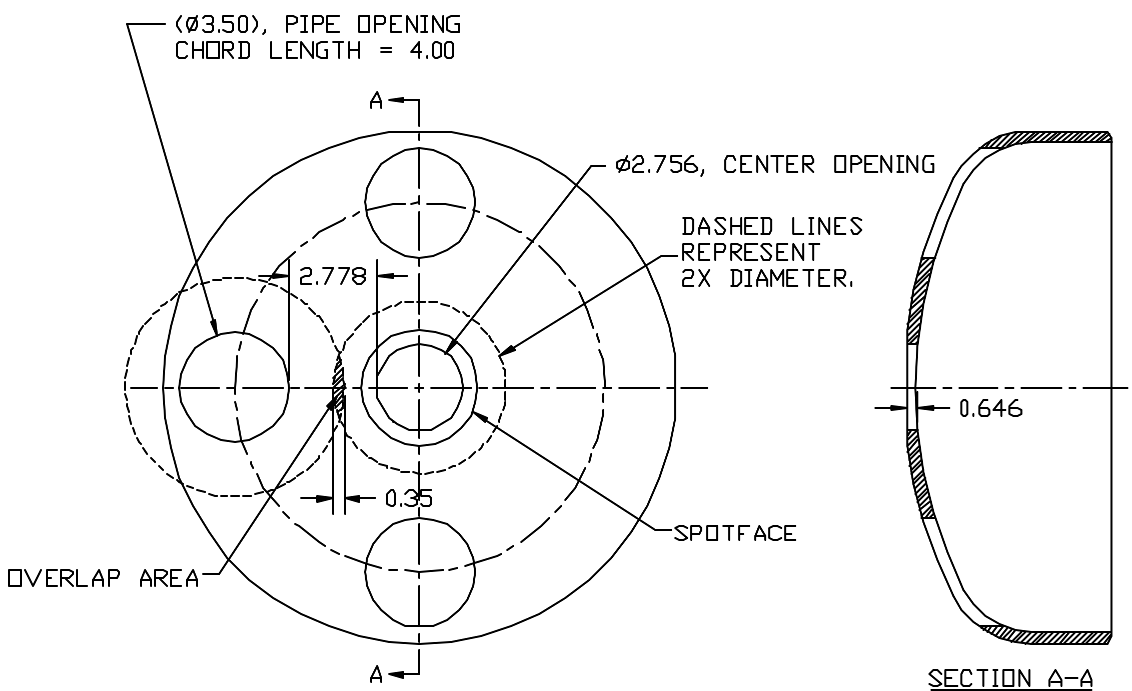

Figure 2.2.1.1 Detail of dome.

Section UG-37 of the Code requires that the minimum area of reinforcement for these openings is:

$$A_r = dt_rF + 2t_nt_rF(1 - f_{r1})$$

where: A_{r1} = area required for center hole

A_{r2} = area required for pipe hole

d_1 = inside diameter of center opening = 2.756 inches

d_2 = inside diameter (chord length) of pipe opening = 4.00 inches

t_r = minimum required thickness of the shell = 0.280 inches

F = correction factor = 1

t_{n1} = nozzle wall thickness for center opening = 0.157 inches

t_{n2} = nozzle wall thickness for pipe opening = 0.065 inches

f_{r1} = strength reduction factor = 1

For this case, $A_{r1} = 0.773$ in^2 and $A_{r2} = 1.121$ in^2.

The area for reinforcement available in the dome is given by the larger of:

$$A_{ac} = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1})$$

or

$$A_{ac} = 2(t + t_n)(E_1t - Ft_r)-2t_n(E_1t - Ft_r)(1 - f_{r1})$$

where: A_{ac1} = calculated required area for the center opening

A_{ac2} = calculated required area for the pipe opening

E_1 = weld efficiency = 1

t = dome thickness = 0.687 inches

t_{co} = dome thickness at spotface, thickness used for center opening = 0.646 inches

The larger values for each area are found to be: $A_{ac1} = 1.008$ in^2 and $A_{ac2} = 1.627$ in^2 from the two expressions above.

It can be seen that these openings are spaced at less than two times their average diameter. Section UG-42 (a) (1) of the Code requires that the available area between openings shall be proportioned between the two openings by the ratio of their diameters. The overlap area is given by:

$$A_{over} = (ratio)\ L_{over}\ (t - t_r)$$

where: A_{over1} = overlap area of the center opening

A_{over2} = overlap area of the pipe opening

L_{over} = length of overlap = 0.35 inches

$ratio_1$ = ratio for center opening = $d_1/(d_1+d_2)$ = 0.41

$ratio_2$ = ratio for pipe opening = $d_2/(d_1+d_2)$ = 0.59

This gives $A_{over1} = 0.06\ in^2$ and $A_{over2} = 0.083\ in^2$. The overlap area from the center opening is subtracted from the available reinforcement area of the pipe opening and vice versa. This leads to the available area for each opening as follows:

$$A_{a1} = A_{ac1} - A_{over2} \qquad \text{and} \qquad A_{a2} = A_{ac2} - A_{over1}$$

This results in the true available reinforcement area for each opening: $A_{a1} = 0.925\ in^2$ and $A_{a2} = 1.567\ in^2$. The available areas are greater than the required areas, so this requirement is met.

Section UG-42 (2) requires that at least 50% of the required area of reinforcement must be between the two openings. The required area between the openings is given by:

$$A_{50\%R} = (A_{r1} + A_{r2})/2 = 0.947\ in^2.$$

The actual area available between openings is given by:

$$A_{between} = L_{between}\,(t - tr)$$

where: $L_{between}$ = distance between openings = 2.778 inches

This gives $A_{between}$ = 1.13 in^2. This requirement is satisfied.

All the requirements of the Code have been satisfied for an internal pressure of 290 psi.

2.2.2 Cylindrical section

The dome consists of the ellipsoidal portion as well as a straight cylindrical section. This can be seen in Figures 2.0.1 and 2.2.1.1. This section is treated as cylindrical shell and the required thickness is given by UG-27 of the Code. The minimum thickness is given by the larger of:

$$t = \frac{PR}{SE - 0.6P}$$

or

$$t = \frac{PR}{2SE + 0.4P}$$

where: P = internal design pressure = 290 psi

R = inside radius of shell = 9.252 inches

S = allowable material stress = 16,000 psi

E = joint efficiency = 0.60

For this case, t = 0.285 inches is the larger value. The shell thickness is 0.394 inches at its minimum so this requirement is satisfied.

The weld between the cylindrical section and the end dome is a single sided butt weld without a backing strip. The maximum load on the weld is due internal pressure. The force due to internal pressure is equal to the cross sectional area in the cylindrical section multiplied by the design pressure. Using the design pressure of 290 psi, this force is equal to 78,000 lbs. The weld is a full penetration weld with a minimum thickness of 10 mm. The stress on the weld is given by

$$t_w = \frac{f_a}{(l)(t_w)}$$

where: τ_w = stress in the weld

f_a = axial force = 78,000 lb

l = linear length of weld = 58.13inches

t_w = weld equivalent thickness = 10 mm/$\sqrt{2}$ = 7.07 mm = 0.278 inch

For this case, the weld stress, τ_w ,is 4820 psi which is below that allowed by UW-15 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.60) = 9{,}600 \text{ psi.}$$

2.2.3 Cold mass pipes

The welds between the cold mass pipes and the end dome are all single sided welds made on the outside of the dome. The maximum load on the weld is a combined load due to the internal pressure and the attached bellows. The force due to internal pressure is equal to the cross sectional area in the pipe multiplied by the design pressure. Using the design pressure of 290 psi, this force is equal to 2,600 lbs. The bellows has an axial spring constant of 68 lb/in and a maximum travel of 1.67 inches. This results in a bellows force of 114 lbs. This force is combined with the force due to internal pressure for a combined force of 2,714 lbs.. As shown in Fermilab drawings 5520-MD-390197 and 5520-MD-390198, these welds are specified to be a 2 mm (0.08 inch) fillet weld. The stress on the weld is given by

$$t_w = \frac{f_a}{(l)(t_w)}$$

where: τ_w = shear stress in the weld

f_a = axial force = 2,714 lb

l = linear length of weld = 11.0 inches

t_w = weld equivalent thickness = 2 mm/√2 = 1.414 mm = 0.056 inch

For this case, the weld stress, τ_w, is 4,406 psi which is below that allowed by UW-15 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.49) = 7{,}840 \text{ psi}$$

The welds between the cold mass pipes and the end flanges are category C lap welds as described in UW-3 (a) (2) and UW-9 (e) of the Code. UW-9(e) requires that the overlap be not less than four times the thickness of the inner plate. In the case of the cold mass pipe, the tube thickness is 0.065 inch. The overlap at the end flanges is 0.67 inch so the requirement is met. The only load acting on this flange is an axial load from the maximum design pressure of 290 psi. The total axial force acting on the flange is 2,600 lb. At the end flange, this force is resisted by the weld between the cold mass pipe and the end flange. As shown on Fermilab drawing 5520-MD-390197, this weld is specified to be a 2 mm (0.08 inch) fillet. The stress on the weld is given by

$$s_w = \frac{f_a}{(l)(t_w)}$$

where: σ_w = stress in the weld

f_a = axial force = 2,600 lb

l = linear length of weld = 11.0 inches

t_w = weld equivalent thickness = 2 mm/√2 = 1.414 mm = 0.056 inch

For this case, the weld stress, σ_w, is 4,221 psi which is below that allowed by UW-18 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.55) = 8{,}800 \text{ psi}$$

2.2.4 Beam tube

The beam tube is inserted through the center of the cold mass. The internal pressure of the cold mass acts as an external pressure on the beam tube. The beam tubes for the LMQXA (Q1) and the LMQXC (Q3) cold masses have different dimensions. The thickness of a shell or tube under external pressure is given by section UG-28 of the Code. Both cases are discussed below. For the Q3 beam tube:

UG-28(c) *Cylindrical Shells and Tubes*. The required minimum thickness of a cylindrical shell or tube under external pressure, either seamless or with longitudinal butt joints, shall be determined by the following procedure.

(1) *Cylinders having D_0/t values ³ 10*:

Step 1. Assume a value for *t* and determine the ratio L/D_0 and D_0/t.

For this case t=1.85 mm, D_0=66.5 mm and L=8.5 m.

Then, L/D_0>50 and D_0/t=35.95.

Step 2. Enter Fig. G in Subpart 3 of Section II, Part D of the Code at the value of L/D_0 determined in Step 1. For values of L/D_0 greater than 50, enter the chart at a value of L/D_0=50.

Step 3. Move horizontally to the line for the value of D_0/t determined by Step 1. Interpolation may be made for intermediate values of D_0/t. From this point of intersection, move vertically downward to determine the value of factor A. From the chart A=0.0009

Step 4. Using the value A calculated in Step 3, enter the applicable material chart in Subpart 3 of Section II, Part D of the Code for the material under consideration. Move vertically to an intersection with the material/temperature line for the design temperature.

Step 5. From the intersection obtained in Step 4, move horizontally to the right and read the value of factor B.

From Fig. HA-2 for 316LN stainless steel, for A=0.0009 and for operation up to 100 F, B=9290.

Step 6. Using the value of B, calculate the value of the maximum allowable external working pressure P_a using the following formula:

$$P_a = \frac{4B}{3(D_o/t)}$$

This gives P_a = 344.6 psi. which is greater than the design pressure of 290 psi. This requirement is satisfied for the Q3 beam tube.

The Q1 beam tube has the following dimensions: t=2.0 mm, D_0=57.0 mm and L=8.4 m. Following the same steps from above for the Q1 beam tube, A = 0.001375 and B = 10,750. This gives P_a = 502.9 psi. This requirement is satisfied for the Q1 beam tube.

The weld between the beam tube and the flange is a category C lap weld as described in UW-3 (a) (2) and UW-9 (e) of the Code. UW-9(e) requires that the overlap be not less than four times the thickness of the inner plate. In the case of the beam tube, the tube thickness is 0.073 inch. The overlap at the end flanges is 0.73 inch so the requirement is met. The only load acting on this flange is an axial load from the maximum design pressure of 290 psi. The total axial force acting on the flange is 174 lb. At the flange, this force is restricted by the weld between the beam tube and the flange and also by the weld between the flange and the end dome. It will be assumed that only one weld is resisting the load and the smaller weld will be chosen. This is the weld between the beam tube and the flange. The stress on the weld is given by

$$s_w = \frac{f_a}{(l)(t_w)}$$

where: σ_w = stress in the weld

f_a = axial force = 174 lb

l = linear length of weld = 7.76 inches

t_w = weld equivalent thickness = 1.8 mm/√2 = 1.27 mm = 0.050 inch

For this case, the weld stress, σ_w, is 450 psi which is below that allowed by UW-18 of the Code given by:

$$(20{,}000\ \text{psi})(0.8)(0.55) = 8{,}800\ \text{psi}$$

2.3 End dome to end ring weld

The end dome to end ring weld conforms to ASME Code, UW-13.2 (d) and is shown in Figure 2.3.1.

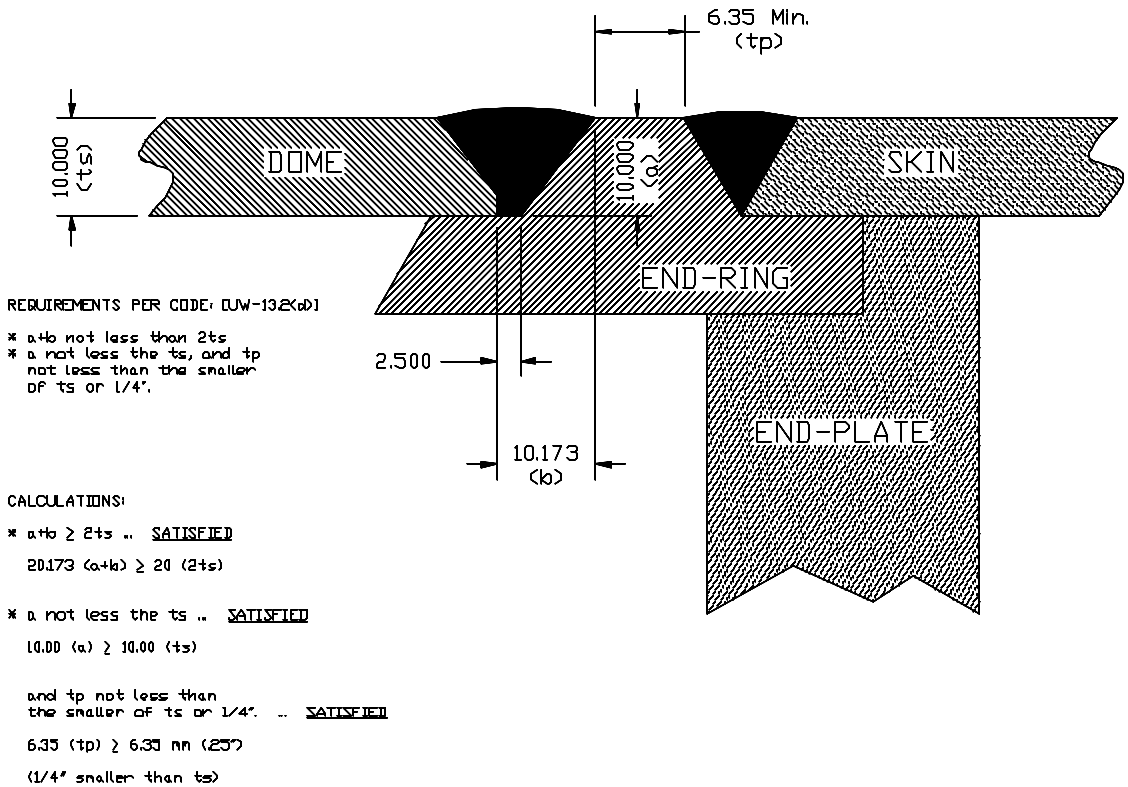

Figure 2.3.1 Detail of cold mass skin to end plate weld.

Using the notation from the figure:

a = 10.00 mm

b = 10.173 mm

t_s = 10.00 mm

t_p = 6.35 mm

UW-13.2 (d) requires that:

(1) $a+b \geq 2t_s$

(2) $a \geq t_s$

(3) $t_p \geq t_s$ or $t_s \geq$ ¼ in (6.35 mm)

All three requirements are satisfied.

2.4 Non-pressure loads

See the KEK cold mass engineering note in Appendix A for the welding, cooldown and gravity load stress analysis.

2.5 Pressure testing

The cold mass assembly will be pressure tested in accordance with Section 5034 of the Fermilab ES&H Manual and UG-100 of the Code. The test pressure is 363 psi, which is 1.25 times the design pressure. The test will be performed after normal working hours and only personnel directly involved with the test will be present. The test medium will be gaseous nitrogen.

2.6 Summary

The LHC cold mass assembly satisfies all the requirements of the ASME Code. It was shown that the MAWP for the cold mass assembly is 290 psi.

Chapter 3

LHC Interaction Region Quadrupole

Cryogenic piping

3.0 Introduction

The cryogenic pipes perform a variety of functions. They transport cryogens down the length of the cryostat during cooldown, warm-up, and in operation. There are nine distinct tube that comprise the cryogenic piping. Their descriptions and a summary of their operating parameters are shown in table 3.0.1. Figure 3.0.1 illustrates all the cryogenic lines in an LHC cryostat.

Table 3.0.1. Cryogenic piping parameters

Description	Fluid	OD (mm)	ID (mm)	P oper (bar)	P max (bar)	T (approx)	Flow (g/s)
Pumping line	Ghe	88.90	85.60	0.016	4.0	1.8 K	8.6
Heat exchanger outer shell	Lhe	168.28	162.74	1.3	20.0	1.9 K	0.0
Heat exchanger inner tube	Lhe	97.54	96.01	0.016	4.0	1.8 K	8.6
Cooldown line	Lhe	44.45	41.96	1.3	20.0	1.9 K	30.0
LHe supply	Lhe	15.88	13.39	0.016	4.0	1.8 K	8.6
4.5K supply and return	Lhe	19.05	15.75	1.3	20.0	4.5 K	1.1
50-70K shield supply	Ghe	38.10	31.75	19.5	22.0	60 K	5.0
50-70K shield return	GHe	38.10	31.75	19.0	22.0	65 K	5.0

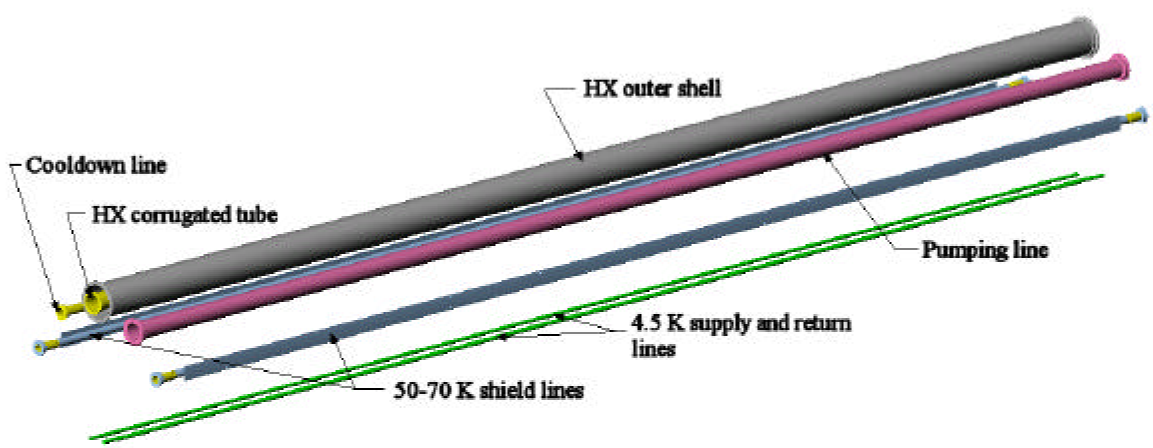

Figure 3.0.1 LHC cryostat cryogenic piping

3.1 Design codes and evaluation criteria

The LHC quadrupole cryostat piping was designed and built, but not inspected, per the requirements of ASME B31.3, "Chemical Plant and Petroleum Refinery Piping". All of the piping welds are made by Fermilab welders certified to the requirements of Section IX of the ASME code, visually inspected as described in B31.3 section 341.4.1(a), and passed a helium leak test per Fermilab engineering specification ES-107240.

3.2 Materials

The pumping line, heat exchanger outer shell, cooldown line, Lhe supply, and 4.5K supply and returns are fabricated from 304 stainless steel. The heat exchanger inner tube is an OFHC copper corrugation. The 50-70K shield supply and return are 6063-T5 aluminum extrusions.

3.3 Pressure loading and analysis

With the exception of the heat exchanger outer shell and inner tube, the minimum thickness is evaluated using the procedures in 304.1.2(a) of ASME B31.3. The minimum tube thickness for seamless or longitudinally welded piping for t<D/6 is given by:

$$t = \frac{PD}{2SE}$$

where: t = wall thickness

P = internal design pressure

D = outside diameter

S = allowable stress from table A-1

E = quality factor from table A-1A or A-1B

Table 3.3.1 summarizes the results of the wall thickness calculation for each of the applicable lines.

Table 3.3.1. Cryogenic piping parameters							
Description	**P (psi)**	**D (in)**	**S (psi)**	**E**	**t req'd (in)**	**P at MTF (psi) ****	**t actual (in)**
Pumping line	59	3.50	20,000	1.0	0.005	1	0.083
Heat exchanger outer shell	na (see below)					100	0.109
Heat exchanger inner tube	na (see below)					1	0.310
Cooldown line	294	1.75	20,000	1.0	0.013	100	0.065
LHe supply	59	0.63	20,000	1.0	0.001	1	0.065
4.5K supply and return	294	0.75	20,000	1.0	0.006	100	0.065
50-70K shield supply	323	1.50	7,300	1.0	0.033	100	0.125
50-70K shield return	323	1.50	7,300	1.0	0.033	100	0.125

**: Relief valve setting at MTF.

In all cases the actual wall thickness of the piping is greater than the minimum required by ASME B31.3. Also in all cases, the maximum pressures at MTF as established by the relief valve settings are less than the design pressures.

3.3.1 Heat exchanger outer shell

The outer shell of the external heat exchanger is a special case when considering the cryogenic piping because it is over 6 inches in diameter, i.e. the diameter above which the boiler and pressure vessel code applies, not the piping code. Application of the Code to determine the minimum required thickness for the outer shell yields the results shown in table 3.3.1.1.

Table 3.3.1.1 Outer shell as a pressure vessel (governing equations (UG-27(c))

$$t = \frac{PR}{SE - 0.6P}(circumferential stress) \text{ or } t = \frac{PR}{2SE + 0.4P}(longitudinal stress)$$

Variable	Value	Units	Descriptions and References
P	300	Psi	Internal design pressure
R	3.125	In	Shell inside radius
S	16000	Psi	Subpart 1, Section II, Part D, Table 1A, derated to 80% of allowed
E	0.70		Weld joint efficiency (Table UW-12)
t(c)	0.085	In	Minimum shell thickness when sized for circumferential stress
t(l)	0.042	In	Minimum shell thickness when sized for longitudinal stress
t	0.085	In	Minimum shell thickness

For this case the minimum wall thickness required is 0.085 inch. The outer shell of the heat exchanger is 6 inch IPS, schedule 5 with an outside diameter of 6.625 inches and a wall thickness of 0.109 inch so the requirement is satisfied.

3.3.1.1 End flanges

The welds between the tube ends and the end flanges are category C lap welds as described in UW-3(a)(2) and UW-9(e) of the Code. UW-9(e) requires that the overlap be not less than four times the thickness of the inner plate. In the case of the heat exchanger outer shell, the tube thickness is 0.109 inch. The overlap at the end flanges is 0.6 inch so the requirement is met. The only load acting on this flange is an axial load from the maximum design pressure of 300 psi. The total axial force acting on the flange is 9,700 lb. At the end flange, this force is resisted by the weld between the outer shell and the end flange. As shown on Fermilab drawing 5520-ME-390002, this weld is specified to be a 3 mm (0.12 inch) fillet. The stress on the weld is given by

$$s_w = \frac{f_a}{(l)(t_w)}$$

where: σ_w = stress in the weld

f_a = axial force = 9,700 lb

l = linear length of weld = 20.8 inches

t_w = weld equivalent thickness = 3 mm/$\sqrt{2}$ = 2.12 mm = 0.084 inch

For this case, the weld stress, σ_w, is 5,550 psi which is below that allowed by UW-18 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.55) = 8{,}800 \text{ psi}$$

3.3.1.2 End caps

The end cap closes the heat exchanger outer shell at one end of the Q1 and at one end of the Q3. There is a hole in the cap where the heat exchanger inner pipe exits. This cap is considered a flat head with a central opening. Since the opening is more than half the diameter of the head, UG-39(c)(1) states that the head shall be designed according to Appendix 14 and related factors in Appendix 2 of the Code. This calculation involves choosing a flange thickness, calculating the resulting stresses and comparing those stresses with various allowable stresses. This is repeated until the calculated stresses are less than the allowable stresses that are given in Appendix 2, Section 2-8 of the Code. Table 3.3.1.2.1 summarizes the iteration process.

Table 3.3.1.2.1. End cap thickness results.

End Cap Thickness (inches)	Minimum Design Factor (must be greater than 1)
0.500	0.734
0.625	0.989
0.750	1.272

The resulting end cap dimensions are as follows:

OD = 6.625 inches

ID = 3.567 inches

Thickness = 0.75 inches

The weld that attaches the end cap to the heat exchanger outer tube conforms to Fig. UW-13.2 sketch (d). All of the requirements of the Code are satisfied.

3.3.1.3 Cold mass connection

The heat exhanger outer shell connects to the cold mass through a short vertical tube. The connection of this tube to the outer shell constitutes an opening in the vessel that potentially needs reinforcement. Section UG-37 of the Code requires that the minimum area of reinforcement for these openings is:

$$A_r = dt_rF + 2t_nt_rF(1 - f_{r1})$$

where: A_r = area required

d = inside diameter of opening = 3.75 inches

t_r = minimum required thickness of the shell at the design pressure computed using UG-27(c)(1) = 0.085 (see table 3.3.1.1)

F = correction factor = 1

t_n = nozzle wall thickness = 15.9 mm = 0.625 inches

f_{r1} = strength reduction factor = 1

For this case, $A_r = 0.319\ in^2$. The area for reinforcement available in the shell is given by the larger of:

$$A_1 = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1})$$

or

$$A_1 = 2(t + t_n)(E_1t - Ft_r)-2t_n(E_1t - Ft_r)(1 - f_{r1})$$

where: E_1 = weld efficiency = 1

t = vessel wall thickness = 0.109 inches

For this case, $A_1 = 0.09\ in^2$ from the two expressions above. The available area in the shell is less than the required area so the reinforcement in the nozzle must be evaluated.

The minimum thickness of the nozzle is given by UG-27 and is the larger of:

$$t_{rn} = \frac{PR_n}{SE - 0.6P}$$

or

$$t_{rn} = \frac{PR_n}{2SE + 0.4P}$$

where: R_n = inside radius of nozzle = 1.875 inches

For this case, t_{rn} = 0.049 inches is the larger value using the first of the two expressions above. The nozzle thickness is 0.625 inches, so this requirement is satisfied.

The reinforcement area available in the nozzle is given by the smaller of:

$$A_2 = 5(t_n - t_{rn})t$$

or

$$A_2 = 5(t_n - t_{rn})t_n$$

For this case, A_2 = 0.314 in^2 from the two expressions above. The total available area of reinforcement is given by:

$$A_{tot} = A_1 + A_2$$

For this case, A_{tot} = 0.404 in^2 which is larger than the required area of reinforcement, so this requirement is satisfied.

This opening is greater than half the ID of the shell so the requirements of Appendix 1-7 of the Code must also be evaluated. Section 1-7(a) states that two-thirds of the required reinforcement shall be within the following limits:

1) Parallel to vessel wall: the larger of three-fourths times the limit in UG-40(b)(1), or equal to the limit in UG-40(b)(2);

2) Normal to vessel wall: the smaller of the limit in UG-40(c)(1), or in UG-40(c)(2).

The requirements from these limits give an envelope of 5.626 inches by 0.273 inches. The nozzle is 5.0 inches by 0.109 inches. All of the available reinforcement is within these limits so this requirement is satisfied.

Section 1-7(b)(2) states that the membrane stress, S_m, shall not exceed the allowable stress, S, and also that the maximum combined membrane stress, S_m, and bending stress, S_b, shall not exceed 1.5S at design conditions. Case B of Fig. 1-7-1 gives the membrane stress:

$$S_m = P\left(\frac{R(R_n + t_n + \sqrt{R_m t}) + R_n(t + \sqrt{R_{nm} t_n})}{A_s}\right)$$

where: R_m = mean radius of shell = 3.258 inches

R_{nm} = mean radius of nozzle neck = 2.188 inches

For this case, S_m = 4,833 psi, which is less than the allowable stress, S = 16,000 psi.

The bending stress is given by:

$$S_b = \frac{Ma}{I}$$

where: M = bending moment = 1,128 in-lbs.

a = distance between neutral axis and inside of vessel wall = 0.519 inches

I = moment of inertia about nuetral axis = 0.032 in^4

For this case, S_b = 18,290 psi. The combined stress, $S_m + S_b$, is equal to 23,123 psi which is lower than 1.5S = 1.5(16,000) = 24,000. This requirement is satisfied.

The cold mass connection tube is attached to the outer shell using a fillet weld. The only load acting on this joint is an axial load from the maximum design pressure of 290 psi. The total axial force acting on the flange is 2,500 lb. The maximum force which can be supported by these welds is given by:

$$F = s_w p d t_w E$$

where: F = maximum allowed force in the weld

σ_w = stress in the weld = (20,000 psi)(0.8) = 16,000 psi

d = effective diameter of the weld = 3.5 inches

t_w = weld equivalent thickness = 3 mm/$\sqrt{2}$ = 2.12 mm = 0.084 inch

E = weld efficiency = 49% (per UW-15)

For this case, F = 7,240 lb. which is less than the total force of 2,500 lb. so the weld is sufficient.

3.3.2 Heat exchanger inner tube

The inner tube of the external heat exchager is an OFHC copper corrugated tube. The dimensions are shown on Fermilab drawing 5520-MC-390011. Copper is required for thermal conductivity. The corrugations provide some reservoir for liquid and add to the structural strength of the tube. The pressure loading is shown in table 3.0.1. The most significant load is an external pressure at 20 bar or 300 psi.

The piping code doesn't explicitly address corrugated tubes. It addresses metal expansion joints, but this tube doesn't fall into that category. As a result, a finite element model of the tube was created and subjected to 60 psi internal and 300 psi external pressue loads. The results from these two analyses are shown in figures 3.3.2.1 and 3.3.2.2.

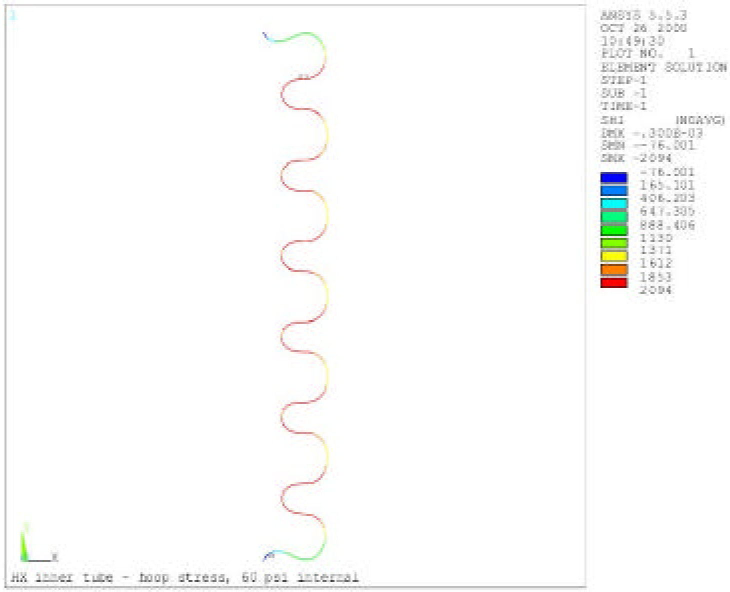

Figure 3.3.2.1 Heat exchanger inner tube hoop stress at 60 psi internal pressure

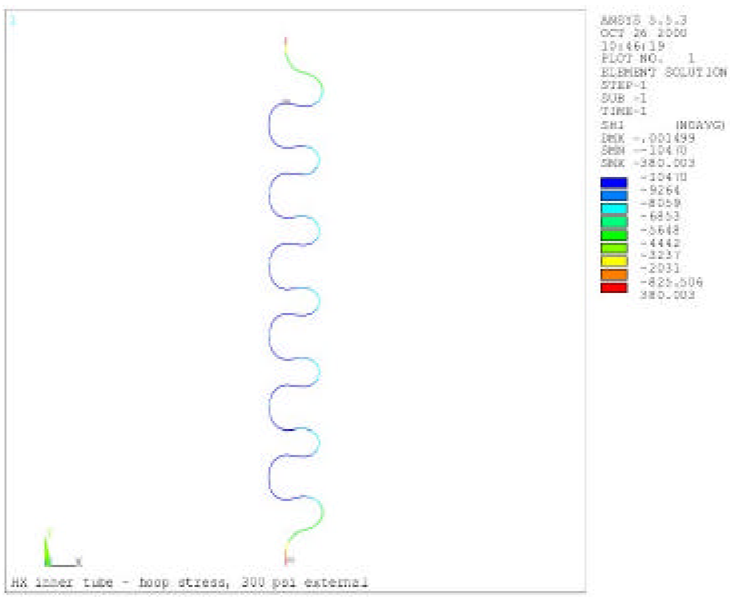

Figure 3.3.2.2 Heat exchanger inner tube hoop stress at 300 psi external pressure

In addition a sample of the inner tube was subjected to a hydrostatic test in a fixture made specifically for that purpose. The sample was tested to 500 psi external pressure with no visible distortion of the convolutions.

3.3.3 Collection volume

The collection volume is a helium overflow tank that connects to the pumping line on the IP end of Q1. The pressure rating of the collection volume is the same as the pumping line, 59 psi. The collection volume is over 6 inches in diameter so this item will be treated as a pressure vessel.

The collection volume consists of a straight section and two domes. The welds that connect the domes to the straight section are category A & B welds as described in section UW-3(a)(1) & (2) and shown in table UW-12 of the Code. They are single-welded butt joints without a backing strip so the joint efficiency, E, is 0.60 for the case where no radiographic examination is made.

The straight section is treated as cylindrical shell and the required thickness is given by UG-27 of the Code. The minimum thickness is given by the larger of:

$$t = \frac{PR}{SE - 0.6P}$$

or

$$t = \frac{PR}{2SE + 0.4P}$$

where: P = internal design pressure = 60 psi

R = inside radius of shell = 4.204 inches

S = allowable material stress = 16,000 psi

E = joint efficiency = 0.60

For this case, t = 0.026 inches is the larger value. The shell thickness is 0.109 inches so this requirement is satisfied.

The domes are ellipsoidal in shape and can be considered an ellipsoidal head. The minimum required thickness is given by UG-32 (d)

$$t = \frac{PD}{2SE - 0.2P}$$

where:

t = minimum required thickness of head after forming, inches

P = internal design pressure = 60 psi

D = inside length of the major axis (ID) = 8.407 inches

S = allowable material stress = 16,000 psi

E = joint efficiency = 0.60

For this case, t = 0.026 inches is the larger value. The dome thickness is 0.109 inches so this requirement is satisfied.

There are several small openings in the domes and shell. UG-36(c)(3)(a) states the openings not larger than 3.5 inches in shells or heads 3/8 inches or less in thickness do not require reinforcement. Also, UG-36(c)(3)(c) states that no two isolated unreinforced openings shall have their centers closer to each other than the sum of their diameters. Both of these requirements are met for all of the openings in the collection volume.

3.4 Summary

The LHC prototype cryostat cryogenic piping satisfies all requirements of the ASME Boiler and Pressure Vessel Code, ASME B31.3, and the Fermilab ES&H manual with the exception of the heat exchanger inner tube which is not explicitly addressed by the codes. However, in the case of this tube, analysis and test results indicate that the integrity of this tube is not compromised by any operating mode at MTF or under LHC operating conditions.

Chapter 4

LHC Interaction Region Quadrupole

Vacuum vessel

4.0 Introduction

The functions of the vacuum vessel are to contain the magnet's insulating vacuum and to provide the structural support of the magnet, shield, and internal piping to the accelerator tunnel floor. In operation, the vacuum vessel is pressurized externally with a differential pressure of one atmosphere. In the event of an internal piping failure the vessel may become pressurized internally. The maximum allowable working pressure is determined in this chapter for both internal and external pressure loading.

4.1 Design codes and evaluation criteria

The LHC quadrupole cryostat vacuum vessel must satisfy all the requirements of the "Vacuum Vessel Safety" section (section 5033) of the Fermilab ES&H Manual. This section states that adherence to the Code is not required, but the design rules may be applied. Because the vacuum vessel contains cryogen lines, the potential for pressurization does exist. If one of these lines were to fail, cryogens could expand to pressurize the vessel to the vacuum system relief valve pressure of 1 psi. Both the Code and the ES&H Manual say that a vessel with an internal pressure of 15 psi or less is not considered a pressure vessel. Therefore, for the purposes of testing at MTF, the vessel functions strictly as a vacuum vessel.

4.2 Materials

The production vacuum vessel shells are fabricated from spiral-welded L 485 MB carbon steel per DIN standard EN 10208-2. This material meets or exceeds the requirements in Fermilab specification 5520-ES-390105 for strength, low-temperature toughness, weldability, and leak-tightness. Material certifications are included in Appendix C. This is a European-standard material and not referenced with Code-allowed materials. However, it is very similar to SA-516, grade 70 that was used for the LHC IRQ prototype vacuum vessel so we use properties for that material here. It was chosen for use at CERN due to its excellent low-temperature toughness properties. Table 4.2.1 summarizes the material composition and physical properties of the two materials. For SA-516, grade 70 the allowed stress is 20,000 psi and the allowable temperature range at this allowed stress is -20 to +500 °F (Section II, Part D, Subpart 1, Table 1A). Section 5031 of the Fermilab ES&H Manual requires derating of the allowable stress to 80% of the allowed value in cases where the vessel is either fabricated in-house or is not code-stamped. This reduces the allowed stress in pressure vessel calculations to 16,000 psi and corresponds to a safety factor of 5. Flanges and access ports are fabricated from 304 stainless steel.

Table 4.2.1: Summary comparison of SA-516 grade 70 and L 485 MB carbon steels

	C (% max)	Mn (% max)	P (% max)	S (% max)	Min yield (ksi)	Min tensile (ksi)	Elongation (% min)
SA-516	0.31	1.2	0.035	0.035	38	70	17
L 485 MB	0.16	1.7	0.025	0.02	70.4	82.7	18

4.3 Structural loading and analysis

The mechanical load on the vacuum vessel consists of the gravity load of the internal components and the vessel itself, the internal radial vacuum load, and the axial vacuum load. The weight of each LHC cold mass is different so for the structural load due to gravity we will consider the weight of the heaviest assembly per unit length, Q3. The Q3 cold mass and internal components weigh 23,500 lb (10,680 kg) and are supported at two places along the length of the vacuum vessel. The radial vacuum load is equivalent to one atmosphere external pressure. The axial load is equal to the cross sectional area of the vessel times one atmosphere pressure or 15,000 lb (6,820 kg). Figure 4.3.1 illustrates a typical LHC IRQ quadrupole cryostat vacuum vessel. Attachments to the accelerator tunnel floor and the internal cold assembly are coincident and occur at the two reinforced sections. The end rings on either end of the vessel provide attachment points for vacuum bellows at one end and the turnaround can at MTF at the other. The four lugs shown at each end ring provide means for securing the vessel to the feedbox and turnaround can.

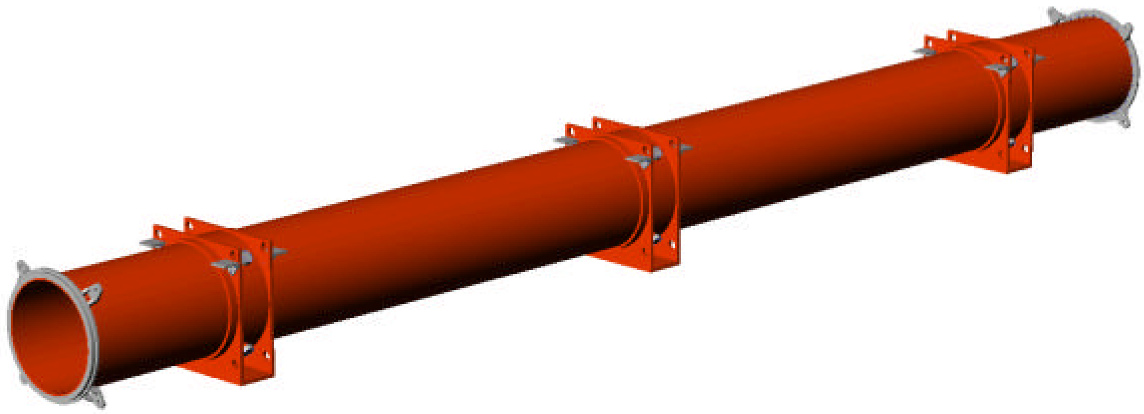

Figure 4.3.1 Typical LHC IRQ cryostat vacuum vessel

The stresses due to the gravity and vacuum loads were determined using a finite element model of the entire assembly. Figure 4.3.2 illustrates the finite element mesh. Gravity acts on the entire assembly. The vacuum loads are applied as a pressure of 15 psi acting inward on the outer vessel wall and as discreet forces acting along the length of the vessel and applied at the end ring.

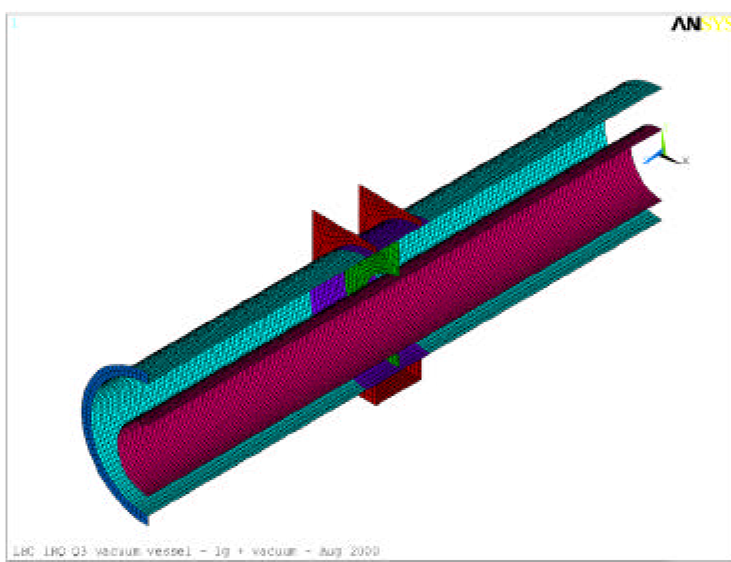

Figure 4.3.2 Finite element mesh from structural and vacuum load analysis

The stresses in the vacuum vessel wall resulting from these combined loads are shown in figure 4.3.3. The stress component displayed is the von Mises equivalent stress that is a combination of principal and shear stress components. It is commonly used to indicate the state of stress in structures that might be indicative of material yielding or failure. The maximum stress in the vessel shell from all the combined loads is 2335 psi and is a bending stress that occurs at the end of the vessel where the end ring attaches. This stress is below the allowed stress in the vacuum vessel material of 16,000 psi.

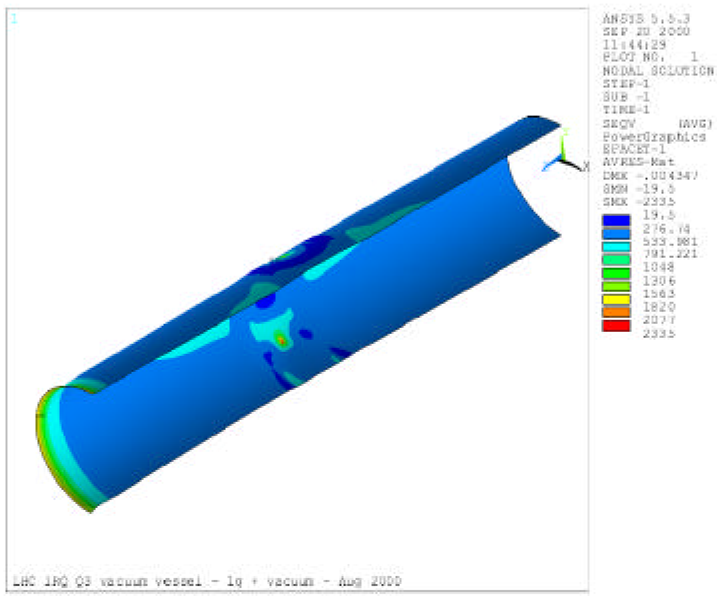

Figure 4.3.3 von Mises stress plot of the vacuum vessel shell only

4.4 Pressure loading and analysis

The vacuum vessel is fabricated in sections to allow adjustment of the individual pieces in an attempt to make as straight a vessel as possible. The individual tube sections are rolled and welded using full-penetration as shown in Fermilab drawing 5520-MD-390131. This drawing is typical of all vacuum vessel tube sections. The spiral-wound longitudinal seam weld is a double butt joint as described in UW-3(a)(1) and shown in table UW-12(1) of the Code. The weld joint efficiency, E, is 0.7 for the case where no radiographic examination is made.

Tables 4.4.1 through 4.4.3 below summarize the Code calculations for the vacuum vessel as an externally pressurized vessel with 1 atmosphere external pressure and as a pressure vessel with 2 atmospheres internal pressure. The interconnecting sleeves and mounting frames are treated as stiffeners. The vacuum vessel has a pressure relief located on the MTF feedbox which opens just above atmospheric pressure. For the sake of the vacuum vessel acting as a pressure vessel however (table 4.4.3) the design pressure is defined to be 2 atmospheres.

Table 4.4.1 Shell as a vacuum vessel (governing equations (UG-28(c))

$$P_a = \frac{4B}{3(D_o/t)}(method1)\ or\ P_a = \frac{2AE}{3(D_o/t)}(method2)$$

Variable	Value	Units	Descriptions and References
Do	36.00	in	Vacuum vessel OD
L(total)	488.00	in	Total shell length
n	3		Number of stiffening rings
L	178.00		Distance between stiffeners
t	0.500	in	Vacuum vessel thickness
E	3.00E+07	psi	Young's modulus
L/Do	4.94		
Do/t	72		
A	0.0004		Subpart 3, Section II, Part D, Figure G.
B	5800		Subpart 3, Section II, Part D, Figure CS-1.
Pa (method 1)	107.41	psi	Calculated maximum allowable external working pressure
Pa (method 2)	111.11	psi	Calculated maximum allowable external working pressure

Table 4.4.2 Stiffening rings (governing equations (UG-29(a))

$$I_s = \frac{D_o^2 L_s (t + A_s/L_s) A}{14}\ and\ B = 3/4\left(\frac{PD_o}{t + A_s/L_s}\right)$$

Variable	Value	Units	Descriptions and References
P	15	psi	External design pressure (1 atm per FESHM 5033)
Do	36.00	in	Vacuum vessel OD
Ls	178.00	in	Distance between stiffeners
t	0.500	in	Vacuum vessel thickness
As	29.528	in2	Assumed cross sectional area (38" OD, 1-1/2" wall, 19.685" long)
B	608		UG-29, Step 1
A	4.05E-05		2 * B / E per UG-29 Step 5
Is	0.445	in4	Required stiffener I

Table 4.4.3 Shell as a pressure vessel (governing equations (UG-27(c))

$$t = \frac{PR}{SE - 0.6P}(circumferential stress) \text{ or } t = \frac{PR}{2SE + 0.4P}(longitudinal stress)$$

Variable	Value	Units	Descriptions and References
P	15	psi	Internal design pressure
R	17.500	in	Shell inside radius
S	16000	psi	Subpart 1, Section II, Part D, Table 1A, derated to 80% of allowed
E	0.70		Weld joint efficiency (Table UW-12)
t(c)	0.023	in	Minimum shell thickness when sized for circumferential stress
t(l)	0.012	in	Minimum shell thickness when sized for longitudinal stress
t	0.023	in	Minimum shell thickness

From table 4.4.1, the maximum allowable external working pressure of the vacuum vessel, Pa, is 107 psi. The minimum pressure required by the Fermilab ES&H manual, chapter 5033 is 1 atmosphere or 15 psi so the requirement is met. From table 4.4.2, the required section modulus of stiffeners is 0.445 in^4. The section modulus of the connecting rings is 5.5 in^4 so the requirement is met. Finally, from table 4.4.3, the minimum shell thickness for the vacuum vessel is 0.023 inch. The vessel wall is actually 0.5 inch so the requirement is met.

4.4.1 Connecting rings

The welds between the individual tube sections and the interconnecting sleeves are category C lap welds as described in UW-3(a)(2) and UW-9(e) of the Code. UW-9(e) requires that the overlap be not less than four times the thickness of the inner plate. In the case of the LHC vacuum vessels, the inner tube thickness is 1/2 inch. The overlap is 2 inches so the requirement is met. From the finite element analysis, the maximum stress in the tube section at the interconnecting sleeve is approximately 775 psi. Since the weld is not explicitly included in the finite element model it is necessary to scale the stress at the weld area by the ratio of the minimum thickness of the weld and the tube thickess. This gives:

$$s_w = s_t \frac{t_t}{t_w}$$

where: σ_w = stress in the weld

σ_t = stress in the tube = 775 psi (from finite element analysis)

t_t = tube thickness = 0.5 inch

t_w = weld equivalent thickness = 6 mm/√2 = 4.24 mm = 0.17 inch

For this case, the weld stress, σ_w, is 2,280 psi which is below that allowed by UW-18 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.55) = 8{,}800 \text{psi}$$

4.4.2 End flanges

The welds between the tube ends and the end flanges are category C laps weld as described in UW-3(a)(2) and UW-9(e) of the Code. UW-9(e) requires that the overlap be not less than four times the thickness of the inner plate. In the case of the LHC vacuum vessels, the inner tube thickness is 1/2 inch. The overlap at the end flanges is only 1 inch so the requirement is not met. The only load acting on this flange is an axial load from the internal vacuum. The total axial force acting on the flange is 15,000 lb. At the end flange, this force is resisted by the weld between the vacuum vessel tube and the end flange. As shown on Fermilab drawing 5520-ME-390129, this weld is specified to be a 6 mm (0.24 inch) fillet. The stress on the weld is given by

$$s_w = \frac{f_a}{(l)(t_w)}$$

where: σ_w = stress in the weld

f_a = axial force = 15,000 lb

l = linear length of weld = 113 inches

t_w = weld equivalent thickness = 6 mm/√2 = 4.24 mm = 0.17 inch

For this case, the weld stress, σ_w, is 780 psi which is below that allowed by UW-18 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.55) = 8{,}800 \text{ psi}$$

4.4.3 Access ports

The access ports in the connecting rings are openings in the vessel. Section UG-37 of the Code requires that the minimum area of reinforcement for these openings is:

$$A_r = dt_rF + 2t_nt_rF(1 - f_{r1})$$

where: A_r = area required

d = inside diameter of opening = 3 inches

t_r = minimum required thickness of the shell at the design pressure computed using UG-27(c)(1) = 0.048 (see table 4.4.3)

F = correction factor = 1

t_n = nozzle wall thickness = 12 mm = 0.47 inches

f_{r1} = strength reduction factor = 1

For this case, $A_r = 0.14\ \text{in}^2$. The area for reinforcement available in the shell is given by the larger of:

$$A_1 = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1})$$

or

$$A_1 = 2(t + t_n)(E_1t - Ft_r)-2t_n(E_1t - Ft_r)(1 - f_{r1})$$

where: E_1 = weld efficiency = 1

t = vessel wall thickness = 1.5 inches

For this case, $A_1 = 5.72$ in^2 from the two expressions above. Since the available area, A_1 is greater than the required are A_r, no additional reinforcement is necessary.

The access ports are attached to the vacuum vessel using fillet welds. These welds support the structural weight of the internal magnet assembly and all other internal components. The weld supports this weight in shear. The maximum force which can be supported by these welds is given by:

$$F = s_w p d t_w E$$

where: F = maximum allowed force in the weld

σ_w = stress in the weld = (20,000 psi)(0.8) = 16,000 psi

d = effective diameter of the weld = 3 inches

t_w = weld equivalent thickness = 5 mm/$\sqrt{2}$ = 3.54 mm = 0.14 inch

E = weld efficiency = 49% (per UW-15)

For this case, F = 10,344 lb. The largest load in any LHC cold mass is 23,500 lb shared by 8 of these welds. In that case each weld supports 2,938 lb so the weld is sufficient.

4.5 Summary

The LHC prototype vacuum vessel satisfies all requirements of the ASME Code and the Fermilab ES&H manual with the exception of the joint between the end rings and the vessel shell. The code requires this lap joint to have minimum overlap of four times the vessel thickness or 2 inches. The design overlap is 1 inch. As shown in 4.4.2 stress in the weld between the end ring and vessel shell is less than 10% of the allowable stress. Since section 5033 of the Fermilab ES&H manual does not require strict adherence to the Code and analysis confirms stresses lower than allowed for the material, exceptional vessel status is not required for the vacuum vessel.

Chapter 5

LHC Interaction Region Quadrupole

Interconnect

5.0 Introduction

The interconnect is the region between the magnet and the feedbox when installed at MTF. The purpose is to transport cryogens, electrical wiring and insulating vacuum from the feedbox to the magnet. There are thirteen total bellows which make up the interconnect consisting of seven unique designs. Their descriptions and a summary of their operating parameters are shown in table 5.0.1.

Table 5.0.1. Bellows operating parameters

Parameter	HX Outer Shell	Cooldown Line	50-70 K Shield	Pumping Line	Cold Mass	MTF Beam Tube	Vacuum Vessel
Internal Media	Lhe	Lhe	Ghe	Lhe	Lhe	Vacuum	Vacuum
External Media	Vacuum	Vacuum	Vacuum	Vacuum	Vacuum	Vacuum	Air
Operating pressure	1.3 bar	1.3 bar	19.5 bar	1.3 bar	1.3 bar	Vacuum	Vacuum
Internal Design Pressure	20.0 bar	20.0 bar	22.0 bar	20.0 bar	20.0 bar	Vac.-1 bar	Vac.-2 bar
External Design Pressure	1 bar	1 bar	1 bar	1 bar	1 bar	Vac.-1 bar	1 bar
Temperature Range	1.9 - 300 K	1.9 - 300 K	50 - 300 K	1.9 - 300 K	1.9 - 300 K	1.9 - 300 K	300 K
Minimum Cycle Life	5000 cycles	500 cycles	500 cycles	5000 cycles	1000 cycles	5000 Cycles	5000 cycles

5.1 Design codes and evaluation criteria

The LHC quadrupole bellows are designed according to the standards of the Expansion Joint Manufactures Association, Inc. (EJMA). All applicable requirements of the Fermilab ES&H manual as well as the ASME Code must also be satisfied.

5.2 Materials

The convolutions on all of the bellows are 316L stainless steel. All other components that make up a bellows assembly are either 304 or 316 series stainless steel.

5.3 Bellows design

The bellows fall into two categories: formed convolutions and welded convolutions. There are six formed and one welded bellows design. Five of the six formed bellows designs are similar and are discussed in section 5.3.1. The other formed bellows, the vacuum vessel bellows, has two sets of convolutions and is discussed in section 5.3.2. The cold mass bellows is the only welded bellows and is discussed in section 5.3.3. All bellows, with the exception of the vacuum vessel bellows, will be fitted with squirm protectors any time there is pressure applied to the bellows and when installed at MTF. In addition, the heat exchanger outer shell, cooldown line, pumping line and 50-70 K shield bellows have integral liners to guard against failure due to elastic instability.

5.3.1 Interconnect bellows

The interconnect bellows are all similar in design, they have one set of convolutions. A typical design is shown in Figure 5.3.1.

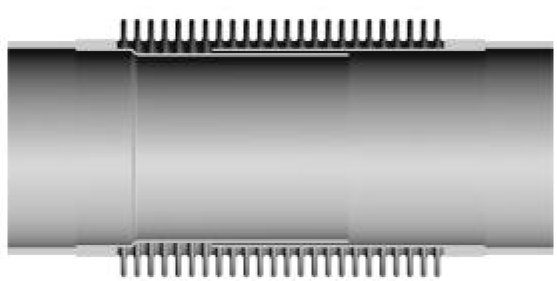

Figure 5.3.1. Typical hydroformed bellows design (cross section).

The convolutions were designed according to EJMA Section C-5.2.2. A computer program was used for the calculations. The input parameters and the results are summarized in Table 5.3.1.1.

Table 5.3.1.1. Interconnect bellows input parameters and results.

	HX Outer Shell	**Cooldown Line**	**50-70 K Shield**	**Pumping Line**	**MTF Beam Tube**
Input					
Bellows ID, in.	7.00	2.25	2.25	4.00	3.00
Number of plys	3	3	3	2	1
Ply thickness, in.	0.014	0.012	0.012	.010	0.008
No. of convolutions	36	8	8	16	18
Convolution pitch, in.	0.222	0.375	.375	0.250	0.250
Convolution depth, in.	0.275	.375	0.375	.375	0.250
Design Pressure, psi	300	325	325	60	15
Travel, in	1.25	1.25	1.25	1.25	0.875
Elastic Modulus, psi	2.83E+07	2.83E+07	2.83E+07	2.83E+07	2.83E+07
Results					
Calc. stress, S1, psi	14,609	5,556	5,556	2,394	588
Calc. stress, S2, psi	10,227	7,333	7,333	2,578	1,490
Calc. stress, S3, psi	1,004	1,841	1,841	590	244
Calc. stress, S4, psi	14,577	34,554	34,554	17,110	4,931
Fatigue cycles	5,298	587	587	9,989	18,144
Axial spring rate, lb/in.	2,232	977	977	301	165
Squirm pressure, psi	304	348	348	80	39

EJMA requires that S1 and S2 be less than the allowable material stress and that (S3+S4) be less than 3 times the allowable material stress. The allowable material stress in this case is 20,000 psi. This requirement is satisfied.

5.3.2 Vacuum vessel bellows

The vacuum vessel bellows consists of two sets of convolutions with a straight section in-between. The convolutions were designed according to EJMA Section C-5.2.2. A computer program was used for the calculations. The input parameters and the results are shown in Table 5.3.2.1.

Table 5.3.2.1. Vacuum Vessel bellows input parameters and results.

	Vacuum Vessel Bellows
Input	
Bellows ID, in.	40.25
Number of plys	1
Ply thickness, in.	0.018
No. of convolutions	8
Convolution pitch, in.	0.500
Convolution depth, in.	1.000
Design Pressure, psi	30
Travel, in	0.7
Elastic Modulus, psi	2.83E+07
Results	
Calc. stress, S2, psi	7619.6
Calc. stress, S3, psi	843.6
Calc. stress, S4, psi	39638.0
Fatigue cycles	1421083.0
Axial spring rate, lb/in.	1006.1
Squirm pressure, psi	237.0

All of the EJMA requirements are satisfied.

The straight section between convolutions is fabricated from 304 stainless steel and is considered a tube or shell under external pressure. Tables 5.3.2.2 and 5.3.2.3 below summarize the Code calculations for the straight section as an externally pressurized vessel with 1 atmosphere external pressure and as a pressure vessel with 2 atmospheres internal pressure. The MTF feedbox, to which this bellows is attached, has a pressure relief that opens just above atmospheric pressure. For the sake of the straight section acting as a pressure vessel however (table 5.3.2.3) the design pressure is defined to be 2 atmospheres.

Table 5.3.2.2 Shell as a vacuum vessel (governing equations (UG-28(c))

$$P_a = \frac{4B}{3(D_o/t)} (method 1) \; or \; P_a = \frac{2AE}{3(D_o/t)} (method 2)$$

Variable	Value	Units	Descriptions and References
Do	40.25	in	Vacuum vessel bellows OD
L	16.00	in	Length of shell
t	0.125	in	Vacuum vessel bellows straight section thickness
E	2.83E+07	psi	Young's modulus
L/Do	0.4		
Do/t	322		
A	0.0006		Figure 5-UGO-28.0, Appendix 5
B	7250		Figure 5-UHA-28.1, Appendix 5
Pa (method 1)	30.02	psi	Calculated maximum allowable external working
Pa (method 2)	35.16	psi	Calculated maximum allowable external working

Table 5.3.2.3 Shell as a pressure vessel (governing equations (UG-27(c))

$$t = \frac{PR}{SE - 0.6P} (circumferential stress) \; or \; t = \frac{PR}{2SE + 0.4P} (longitudinal stress)$$

Variable	Value	Units	Descriptions and References
P	30	psi	Internal design pressure
R	20.00	in	Shell inside radius
S	15,040	psi	Section VIII, Division 1, Table UHA-23, derated to 80% of allowed
E	0.60		Weld joint efficiency (Table UW-12)
t(c)	0.067	in	Minimum shell thickness when sized for circumferential stress
t(l)	0.033	in	Minimum shell thickness when sized for longitudinal stress
t	0.067	in	Minimum shell thickness

From table 5.3.2.2, the maximum allowable external working pressure of the vacuum vessel, Pa, is 30.02 psi. The minimum pressure required by the Fermilab ES&H manual, chapter 5033 is 2 atmospheres or 30 psi so the requirement is met. From table 5.3.2.3, the minimum shell thickness for the vacuum vessel bellows straight section is 0.067 inch. The straight section wall is 0.125 inch so the requirement is met.

5.3.3 Cold mass bellows

The cold mass bellows is a welded bellows. Since EJMA only covers convoluted bellows, this bellows is vendor designed per our specifications. These specifications are the operating parameters listed in Table 5.0.1. There is a bellows protector that fits on the OD of the convolutions to protect the bellows from squirm. The bellows protector must be installed prior to any pressurization of the bellows. This can be seen in Figure 5.3.3.1.

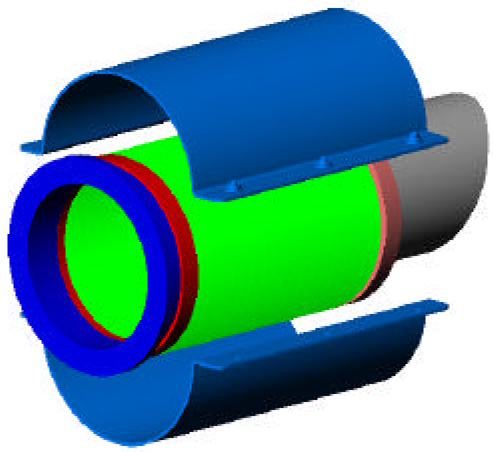

Figure 5.3.3.1. Cold mass bellows protector (exploded view).

5.4 Summary

All of the bellows to be used in the interconnect at MTF for the LHC quadrupoles meet the requirements of EJMA as well as all applicable ASME Codes and the Fermilab ES&H manual.